Evaluation of Coal-Fired Fluid Bed Combined Cycle Power Plant

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ABSTRACT

Recent studies and research indicate that fluidized-bed combustion systems, operating at atmospheric or elevated pressure in a combined cycle power plant, offer the potential for producing electrical energy from coal within present environmental restraints for clean flue gas emissions and at a cost less than for conventional steam power plants utilizing low-sulfur coal or flue gas cleanup equipment. The team of Burns and Roe Industrial Services Corporation, United Technologies Corporation, and Babcock & Wilcox Company is under contract to the Department of Energy to prepare a conceptual design for such a plant. The major objectives of this program are to identify the technology required to develop a coal-fired pressurized fluid bed combustor to drive an industrial gas turbine and to define the technical and economic characteristics of a nominal 600 MW base- or intermediate-load combined cycle power plant.

Several cycle configurations with variations of cycle parameters were investigated during the course of this study. These include the consideration of different pressure ratios, the use of an unfired and fired steam bottoming cycle, and reheating the gas stream before the power turbine. Efficiency estimates for these variations range from about 38 percent for the unfired waste heat system to over 43 percent for the reheat system. As a result of various trade-off studies, a commercial plant cycle arrangement has been selected which incorporates a coal-fired pressurized fluid bed combustor, operating at 10 atm and 1650 F, and supplementary firing of the gas turbine exhaust in a coal-fired atmospheric pressure fluid bed boiler which produces 2400 psig/1000 F/1000 F steam. Preliminary estimates for coal pile to bus bar efficiency for the selected system are around 41 percent (gross, HHV).

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INTRODUCTION

Over the years, coal has become a major source of energy for power generation by electric utilities. However, it has become apparent that the use of coal requires control of the products of combustion to be compatible with the environment. This fact, coupled with the increased emphasis on coal usage has created an incentive to develop alternate methods of extracting energy from coal in an environmentally and economically acceptable way. In addition, it is highly desirable that these alternate methods result in more efficient coal utilization. Recent studies (1) indicate that the use of gas turbines in conjunction with fluidized bed combustion systems in a combined cycle power plant offer potential for satisfying these needs.

The feasibility of burning coal directly in an open cycle gas turbine was investigated by Bituminous Coal Research Inc., as early as 1944 when a coal-fired competitor to the diesel engine for railroad applications was being sought. In the intervening years, a number of organizations have attempted to design and test direct coal-fired gas turbines. However, problems with corrosion, erosion, and deposition on turbine blading due to the amount and nature of the ash passed through the high-temperature combustion zone have prevented the development of a commercially viable product. The low temperatures associated with fluidized bed combustion should alleviate the problems. Under Department of Energy (DOE) sponsorship (Contract EX-76-C-01-2371) the Burns and Roe Industrial Services Corporation, United Technologies Corporation, and the Babcock & Wilcox Company have formed a team to investigate the feasibility of a combined cycle plant utilizing a gas turbine with a pressurized fluid bed combustor. The purpose of this paper is to present important preliminary findings from the first year of effort on the DOE program. Because of the exploratory nature of this effort, some desired technical information is not yet available; indeed, more questions might be raised than are answered by this discussion. Nevertheless, it is deemed appropriate to present the preliminary findings to stimulate early discussion of this promising concept.

AIR-COOLED PRESSURIZED FLUID BED

Fluid bed combustion as currently discussed involves the combustion of coal in a fluid bed containing a crushed sulfur acceptor such as limestone or dolomite. Pressurized fluid bed (PFB) combustion is similar to atmospheric fluid bed (AFB) combustion except that the process takes place under a pressure of several atmospheres such as would exist at the exhaust of the compressor of a gas turbine unit. PFB combustion, therefore, offers the potential of serving as the gas turbine combustor (Figure 1). This use of a PFB as a gas turbine combustor has been studied by several investigators (2 through 7). Indeed, a 1-MW gas turbine has operated with a PFB combustor burning coal (8).

The temperature of the combustion process would be controlled by heat extraction from the bed and/or by controlling the fuel-air ratio in the bed. It would be necessary to maintain the PFB temperature at about 1650 F to minimize the release of volatile alkaline metal compounds which would otherwise cause severe corrosion in the gas turbine and to assure an operating margin below the coal ash softening temperature to prevent agglomeration within the bed. The low combustion temperature also would result in NO_x emissions that are lower than the Federal EPA limits for coal fuel.

Higher PFB operating temperature would be beneficial to cycle performance and carbon utilization. It appears (9 through 13) that fluid beds could be satisfactorily operated up to 1750 F without incurring problems with sulfur retention or ash sintering, but deposits of elutriated material on the walls of the primary cyclone and in the turbine could be excessive. Considering the experience reported in the literature (9), a bed operating temperature of 1650 F was selected for the cycle analysis and PFB combustor design.

As the mechanical design of the PFB combustor developed it was determined that temperatures greater than 1650 F would not be practical. The heat exchange surface within the PFB must be designed for the bed temperature plus a margin for operating variations. Consequently, for the 1650 F bed temperature a 1700 F design temperature was used for the bed internals. At this temperature level the available materials exhibit little strength. The lower allowable stress levels that would result from using higher bed temperature would make such a design impractical, if not impossible. Also, while corrosion of the in-bed surface has not been quantified, it would be expected to be more severe at higher operating temperatures.

With the PFB process it should be possible to capture sufficient sulfur products to permit use of high-sulfur coals and still meet the current EPA limit of 1.2 lb $SO_2/10^6$ Btu input. For a typical 3.4 percent sulfur, 12,000 Btu/lb HHV coal the required sulfur removal efficiency is about 80 percent. Dolomite appears to be an effective sulfur acceptor, and available data (9, 10) indicate that a calcium/sulfur ratio near 1.0 should be adequate to achieve the desired 80 percent sulfur retention at the selected bed operating conditions.

A low fluidizing (superficial) gas velocity is desirable to reduce elutriation from the bed, thereby reducing both the carbon loss and the required particulate cleanup duty. It should be noted that the size of both the coal and dolomite feed must be properly related to the fluidizing velocity, with increased velocity implying increased sizes. Low velocity also implies a larger bed area resulting in a shallower bed and, hence, lower bed pressure loss. A fluidizing velocity of 2.5 - 3.0 fps was selected for the PFB design reflecting previous work (9 through 11).

Even with low fluidizing velocity, a highly efficient particulate removal system would be required to prevent excessive turbine blade erosion. Since the cost of the particulate removal system is strongly influenced by the volume of gas passing through it, one method of reducing the system cost would be to limit the combustion air flow (and hence the dirty gas flow) to only as much as required for coal combustion within the PFB. This could be accomplished by splitting the compressor discharge flow with approximately 25 percent of the air being routed to the PFB combustion zone and the remainder of the air being routed through the bed cooling system consisting of tubes immersed within the fluid bed. The heat released during the combustion process would be transferred to the cooling air less than one-quarter of the oxygen available in the air, the turbine exhaust gas could support considerable firing of additional coal. For this study, an AFB steam generator was considered as the means for capturing the SO_2 released during the final combustion process. The AFB could be steam cooled, as noted in Figure 4, or air cooled either by varying excess air to the bed or by using a split-flow arrangement similar to that described for the PFB in Figure 1. With either air-cooled approach, heat would be recovered from the air and combustion gases in a waste heat steam generator.

The performances of these various combined cycle configurations and of the simple cycle gas turbine are compared in Figure 5. Selected component efficiency, pressure loss, and temperature assumptions used in the calculations are summarized in Table I. As expected, the waste heat recovery system displays the lowest efficiency (about 38 percent, HHV) but, since it requires combustion at only one point in the cycle, it is a less complex configuration than the other cycles and has been utilized as a reference point in the economic analysis. The reheat system offers the highest potential efficiency (43 percent, HHV) but increases the complexity of the gas turbine design and requires a reheat fluid bed combustor with an associated particulate removal system. The PFB gas turbine topping of the AFB steam cycle has an attractive efficiency (approaching 41 percent, HHV) and shows promise for minimum equipment cost because of its relatively high specific work.

ECOMONIC ANALYSIS

The selection of the commercial plant configuration cannot be made on the basis of performance alone. The most important selection criterion is overall cost of electricity; therefore, an order of magnitude analysis was made to estimate the relative capital and operating costs of the alternative configurations. The operating cost differences due to fuel consumption were expressed in terms of equivalent capitalized costs where a one point difference in efficiency would give an equivalent fuel savings of $10/kW.

The results of this economic screening analysis are given in Table II. All costs are given as incremental costs relative to the unfired waste heat recovery system which was taken as the base. The exhaust-fired, steam-cooled AFB with a gas turbine pressure ratio of 10 and the gas turbine system with reheat before the power turbine have the lowest evaluated net relative costs. The cost differential between these two systems is not statistically significant. The power turbine reheat cycle requires a more complex gas turbine design and additional hot particulate removal equipment. In addition, little data is available for design of a PFB combustor at the 2.5 atm pressure existing at the reheat point. Therefore, it was felt that the PFB cycle with an exhaust-fired steam-cooled AFB would offer less technical risk.

Capital costs were not estimated for all major pieces of equipment of systems required in the plant. Table III contains a list of those items which were considered. Obviously, some major systems (such as the coal and sorbent feed systems to the AFB and the low-pressure reheat PFB combustors) were omitted which would tend to decrease the advantage of the reheat and exhaust-fired cycles. However, it was felt that the differences in the costs of these systems would not be large enough to offset the differences shown on Table II. Therefore, there is a strong probability that the trends shown in this study could be confirmed by more detailed design and cost estimates of the alternatives.

It should be recognized that the cost estimates did not consider some of the material, equipment, and other balance of plant costs normally associated with the items indicated on Table III. In addition, little more than conceptual outline drawings were available for many items that were considered. The basic intent of the effort was to provide a systematic approach for summarizing the relative pros and cons of each cycle on the basis of the preliminary design definition that was available. While each pro and con was, in effect, weighted on a cost basis, it would be misleading to consider the numbers shown as anything more than rough order of magnitude.

COMMERCIAL PLANT CONFIGURATION

On the basis of the preceeding screening analysis, the PFB/AFB combined cycle power plant was selected for the commercial plant conceptual design study. During the course of the design study, further optimization of the selected configuration led to incorporation of three stages of regenerative feedwater heating and an adjustment in the relative power split between the gas and steam turbines. The resulting system, illustrated in Figure 6, utilizes two 63.5 MW gas turbines with two PFB combustors per gas turbine. The gas turbines would exhaust into a single exhaust-fired AFB steam generator and carbon burnup bed (CBB) which would generate steam at 2400 psig 1000 F/1000 F to drive a single 461.4 MW steam turbine. The resulting gross plant output would be 588.4 MW. Selected performance and cost data are summarized in the last column of Table II.

The gas turbine assumed for this study is a base load design which represents a modification of UTC's FT50 gas turbine or an engine of similar performance and physical characteristics. It would operate at 10:1 pressure ratio with 1600 F inlet temperature and have all necessary ducting to allow discharge of compressor air to the PFB combustor and return of hot gases to the turbine.

The PFB combustors, depicted in Figure 7, would heat the compressor discharge air from approximately 600 F to 1600 F. The compressor discharge air would enter the bottom of the refractory lined pressure vessel. The combustion air would flow through bubble caps in the distributor plate and into the fluidized bed. The cooling air would flow through supply pipes at the distributor plate to the inlet headers of the cooling circuits, through the tubes, and finally would be collected at the hot air outlet manifold. The flow split between cooling air and combustion air would be controlled by biasing valves in the hot air outlet piping and the hot gas outlet piping. The heat transfer from the bed to the cooling air would require a large surface area and a large bed volume. The desire to maintain a low bed superficial velocity (of the order of 3 ft/sec) is compatible with this large volume and would result in an expanded bed height of approximately 22 ft to submerge the cooling system within the bed.

Incoloy 800 alloy was selected for all material exposed to the fluid bed. This material has had greater usage than the other available high temperature alloys, and its physical properties (forming, welding, etc.) are better established. Also, currently available corrosion, creep, fatigue, and other data indicate that this alloy should give suitable life for the cooling system. However, ultimate material selection must eventually be based on the outcome of other more rigorous investigations of material characteristics within a PFB environment.

Operation at the elevated temperature of the PFB presents significant challenges in designing to accommodate the expected thermal expansion. The air in the bed cooling system would undergo nearly 1000 F temperature change from the inlet to the outlet in a total tube length of less than 26 ft. In addition, the cooling system from the distributor plate to the outlet header must be supported by the pressure vessel which operates at a temperature of 250 F. The design philosophy has been to support the outlet manifold and inlet and outlet headers of the cooling system from the same elevation on the vessel wall and to use U-shaped cooling tubes between the inlet and outlet headers. These U-shaped tubes would be designed with sufficient flexibility to accommodate the differential temperature along the length of the tube.

As previously noted, a particulate removal system would be required to limit the solid loading entering the turbine. Because of lack of actual operating experience with PFB exhaust gases in gas turbines, further testing is required to determine the acceptable level of particulate concentration in the gas entering the turbine. On the basis of limited data (14), an estimate of allowable gas turbine particulate loading was made showing that particles greater than 10 microns in size would give unsatisfactory turbine life, particles less than 2 microns in size would have negligible effect, and that some limited amount of particulate in the 2-10 micron size could be tolerated within the gas turbine. These estimates are compared in the top two lines of Table IV to the estimated particulate loading in the gas exiting from the PFB combustor.

Since the design requirements and characteristics of particulate removal systems are not fully known at this time, two different technologies were investigated in developing the overall plant design. The two concepts are a high-efficiency rotary flow cyclone and a granular bed filter, both of which are in the developmental stage at the temperature, pressure, and size required for the PFB combustion process. From a theoretical standpoint, both types of particulate collectors should meet the requirements of a commercial plant. The estimated effectiveness of the particle collectors is indicated in Table IV where the collector effluent is seen to satisfy the gas turbine requirement. Final dilution of the collector effluent with cooling air which bypassed the PFB combustion zone should reduce the particle concentration well below that required for the gas turbine. Only testing under actual operating conditions will ensure the suitability of these collectors.

The exhaust gases from the two gas turbines would be routed to the AFB steam generator system consisting of four AFB main beds in one structure (Figure 8) and a separate CBB. The main beds would combust coal using the exhaust of the gas turbines as combustion air. Unburned char elutriated from the AFB main beds would be captured and combusted in the CBB. The CBB would be in a separate enclosure, but the steam cooling system would be in parallel with that for the main beds. Most of the superheater surface would be in three of the four beds, with the fourth containing only evaporator surface. The four main beds would each exhaust hot gas upward into a common convection section of the AFB steam generator. All of the reheater tubes and a portion of the primary superheater would be in the convection section. Gas from the convection section would flow into the economizer section. The CBB would consist of three beds, each with two compartments for load turn down control. All boiler surface would be above the beds in the convection zone.

The flue gas from the AFB boiler, after passing through high efficiency multiclones, go through a high temperature electrostatic precipitator. The electrostatic precipitator would be designed for a maximum temperature of 750 F. The total volume of flue gas to be handled by the precipitator is 3.2×10^6 ACFM. The precipitator would have four electric fields in series. The total particulates emission would be less than 0.1 lb per million Btu of heat input. The gas stream from the precipitator would pass through the low level economizer to the induced draft fans and stack.

The hypothetical Middletown, USA site was selected for location of the PFB combined cycle power plant. An area site plan for the prospective power plant is shown in Figure 9. The plant island is centrally located with the cooling tower and switchyard to the east, coal and sorbent storage areas to the south, and wastewater treatment plant to the west.

CONCLUDING REMARKS

The air-cooled PFB offers the potential of using coal-fired gas turbines to top a more conventional coal-fired steam plant. The resulting combined cycle power plant has the capability of more efficient conversion of coal to electricity with the potential of yielding an overall lower cost of electricity than can be obtained with current technology. The PFB system requires development of high efficiency hot particulate removal systems and demonstration of material suitability. However, the technological challenges facing this type of system are less demanding than those for other advanced coal-fired conversion systems presently under study because of the lower temperatures and reduced degree of coal conversion and processing required. In closing, the prospective performance, economic, and environmental advantages of combined cycle power plants using PFB combustors suggest that development of this promising concept be energetically pursued.

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TABLE I

SYSTEM ASSUMPTIONS FOR PERFORMANCE ANALYSIS

Combustion Efficiency, %		
PFB, main and reheat		99.0
AFB		98.5
Pressure Loss, % of local gas pressure		
	Bed	Cooling Tubes
PFB, main and reheat	10.0	10.0 (air)
AFB	9.2	- (steam)
Temperature, ^{o}F		
	Bed	Cooling Tubes
PFB, main	1650	1575 (air)
PFB, reheat	1550	1475 (air)
AFB	1550	- (steam)
Component Efficiency, %		
Electric generator (steam turbine)		98.4
Electric generator (gas turbine)		98.7
Electric motors		95.0
Boiler feed pump		82.0
Boiler feed pump drive turbine		75.0
Condensate pump		82.0
ID fan		70.0

TABLE II

PFB COMBINED CYCLE POWER PLANT COST SUMMARY

	Screening Analysis							Selected Cycle
Cycle Type	Waste Heat		Exhaust-Fired Air-Cooled AFB		Exhaust-Fired Steam-Cooled AFB		Power Turbine Reheat	Exhaust-Fired Steam-Cooled AFB
Gas Turbine Pressure Ratio	10	16	10	16	10	16	16	10
Number of Feedwater Heaters	0	0	0	0	0	0	0	3
Output per Gas Turbine, MW								
Gas Turbine	66.7	62.0	60.0	55.0	63.5	58.4	77.0	63.5
Steam Turbine	32.0	19.0	86.0	87.0	134.0	118.0	88.0	230.7
Total	98.7	81.0	146.0	142.0	197.5	176.4	157.0	294.2
Specific Work, kW-sec/lb-air	127	100	179	174	241	216	192	350
Gross Efficiency, % (HHV)	38.4	37.2	41.0	40.3	40.9	40.4	43.1	41.1
Relative Equipment Cost, $/kW								
Combustion System	Base	+33	-8	-26	-16	-12	-42	-34
Prime Movers & Electrical	Base	+55	-38	-17	-49	-32	-24	-61
Miscellaneous	Base	-5	+76	+79	-10	- 2	+20	-15
Subtotal	Base	+83	+30	+36	-75	-46	-46	-120
Equivalent Fuel Savings, $/kW	Base	+12	-26	-19	-25	-20	-47	-25
Net Relative Cost, $/kW	Base	+95	+ 4	+17	-100	-66	-93	-145

TABLE III

MAJOR EQUIPMENT INCLUDED IN COST SUMMARY

- Main PFB Coal/Sorbent Feed System
- Gas Turbines/Generators
- PFB Main Combustors
- PFB Reheat Combustors
- AFB Combustors
  (Excluding: Flues, Duct, Cyclones, Fans, Coal/Limestone Feed System)
- Electrical Equipment
- Steam Turbine/Generators
- Waste Heat Boilers
- Electrostatic Precipitators

TABLE IV

PARTICLE SIZE DISTRIBUTION AND LOADINGS

Size Range	Predicted Particle Concentration, gr/scf Under 2μ	2-10μ	Over 10μ
Gas Turbine Limit	unlimited	0.01	nil
PFB Effluent	0.3	2.0	6.4
Collector Effluent	0.06	0.01	0.00
Entering Turbine	0.02	0.003	0.000

Fig. 1

PFB Combustor for Gas Turbine

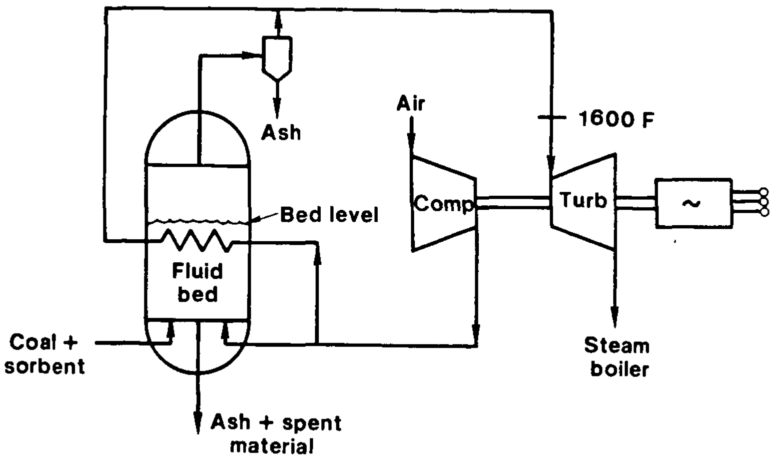

Fig. 2

Unfired Waste Heat Recovery Cycle

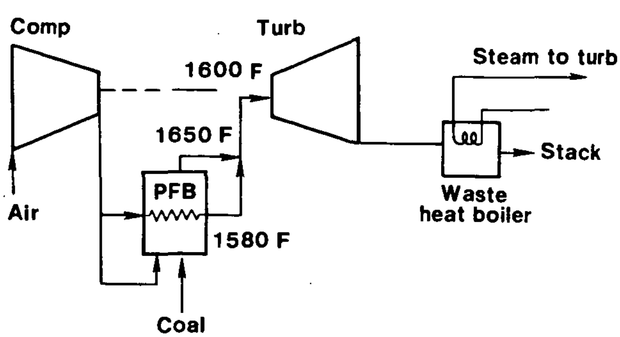

Fig. 3

Power Turbine Reheat Cycle

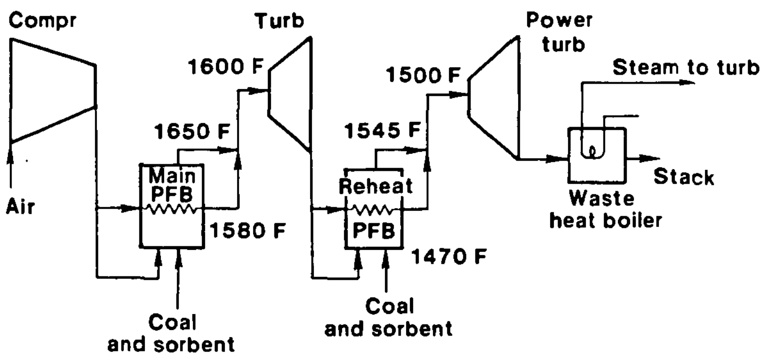

Fig. 4

Exhaust-Fired Steam-Cooled AFB Cycle

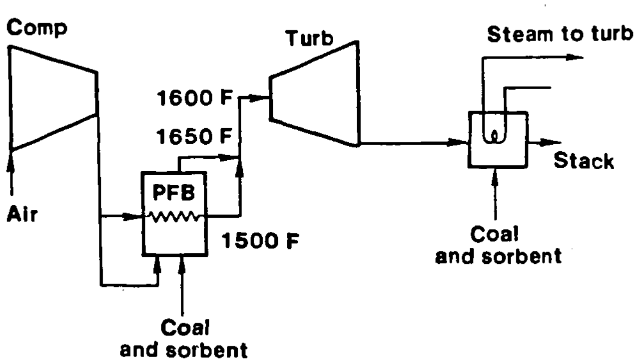

Fig. 5

PFB Cycle Performance Comparison

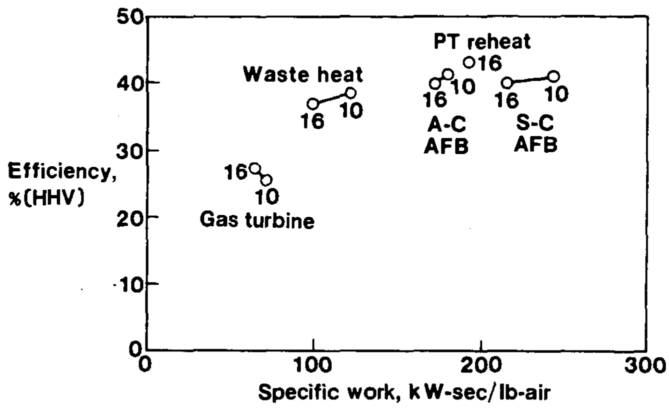

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Fig. 6

PFB/AFB Combined Cycle Power Plant

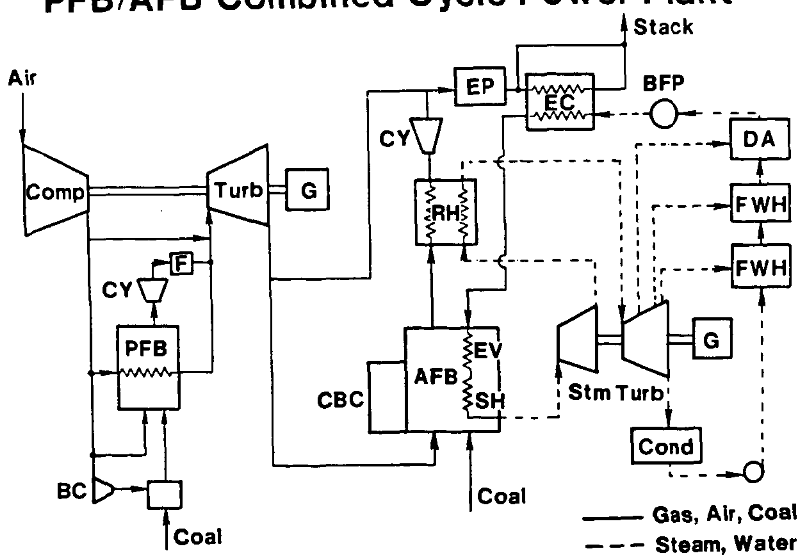

Fig. 7

Pressurized Fluid Bed Combustor

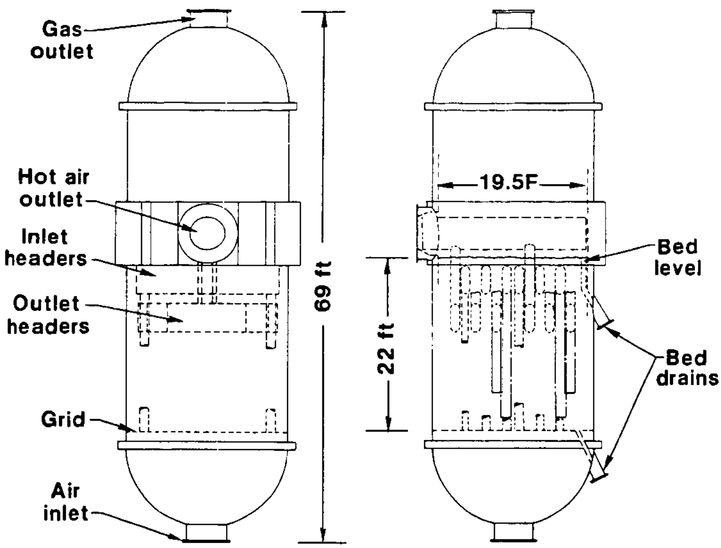

Fig. 8

Atmosphere Fluid Bed Boiler

(Sectional sideview)

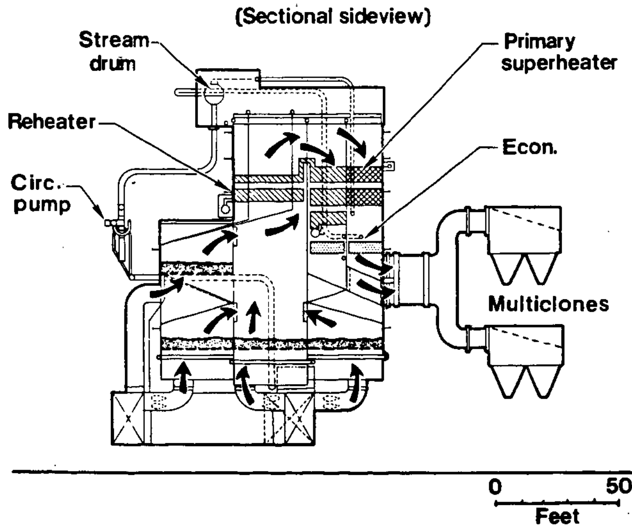

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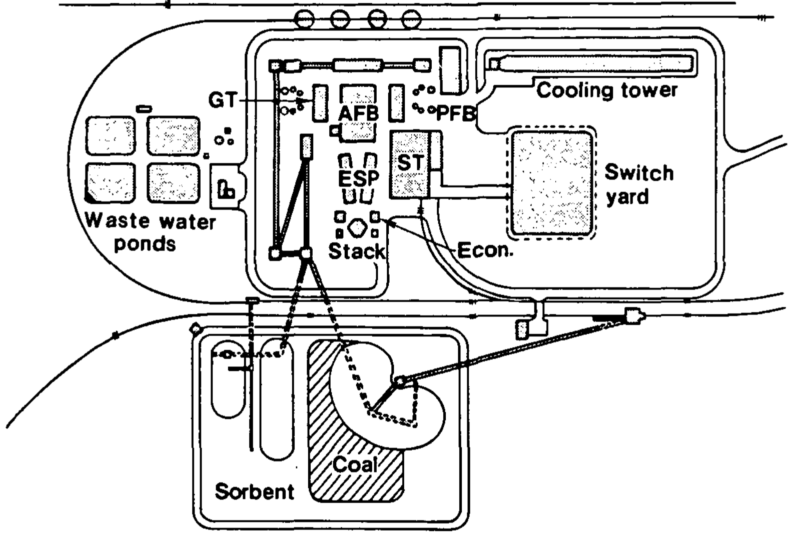
Fig. 9

Plot Plan for PFB Combined Cycle Power Plant



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